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# Numerical analysis on a double pipe heat exchanger with twisted tape induced swirl flow on both sides

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**Abstract**

Twisted tape inserts are widely used for enhancing heat transfer in heat exchangers. They enhance heat transfer by inducing swirl flow in the flow channel, thereby enabling good mixing within the fluid and by increasing the effective flow length of the flow channel. They also increase pressure drop but their overall performance is found to be advantageous in many cases. In this work, an attempt is made to analyse the performance of a modified double pipe heat exchanger with twisted tape induced swirl flow on both sides. The numerical analysis were done in turbulent flow conditions with twisted tape inserts of twist ratio 5 and 3. The results obtained are validated using established correlations available in the literature. The fin effect of twisted tape is also discussed.


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*Keywords:* Double pipe heat exchanger; Flow channel, Twisted tape; Swirl flow; Twist ratio; Annulus; Fin effect

## 1. Introduction

Heat exchangers are one of the most important class of thermal energy handling devices used in industries. Their specific applications can be found in power production, space heating and air conditioning, waste heat recovery, chemical processing etc. Heat exchangers are generally regarded as complex devices. Among them, the double pipe heat exchangers are the simplest type in construction and analysis. But their importance is no way less, since they are widely used in industries.

The need to improve the thermal performance of heat exchangers have led to many modifications on them so as to affect energy, material cost savings as well as consequential mitigation of environmental degradation. These


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methods are referred to as heat transfer enhancement or heat transfer augmentation techniques. Enhancement techniques, reduce the thermal resistance in a conventional heat exchanger by promoting higher convective heat transfer coefficient with or without increase in surface area. As a result, the size of a heat exchanger can be reduced or the heat duty of an existing heat exchanger can be increased. The various heat transfer enhancement techniques can be classified as "passive" and "active" techniques. Passive techniques do not require direct input of external power. They generally use surface or geometrical modifications or incorporate an insert material or additional device. In the case of active techniques, some form of external power is applied to achieve the desired flow modification and improvement in the rate of heat transfer. Twisted tape (TT) inserts are one of the most important passive heat transfer enhancement methods used in circular channels. It is a swirl flow device. When they are inserted in circular channels, swirl flow is imparted to the fluid. The enhancement in heat transfer is due to the agitation of fluid, increase in effective flow length and mixing induced by cross stream secondary flows. Twisted tapes are identified by a parameter called "Twist ratio" usually denoted by *y* and is given by

$$y = {}^{H}\!/_{d} \qquad (1)$$

Fig.1 shows the characteristic dimensions of twisted tapes.

**Nomenclature**

| | |
|---|---|
| $C_p$ | Specific heat, J/kg K |
| $D_h$ | Hydraulic diameter, m |
| $d$ | Diameter of the tube, m |
| $f$ | Friction factor |
| $f_a$ | Friction factor associated with modified heat exchanger |
| $f_0$ | Friction factor associated with ordinary heat exchanger |
| $H$ | Half pitch of twisted tape, m |
| $h_a$ | Average heat transfer coefficient, W/m$^2$ K |
| $h_l$ | Local heat transfer coefficient, W/m$^2$ K |
| $k$ | Thermal conductivity, W/m K |
| $L$ | Length of flow channel, m |
| $Nu$ | Nusselt number |
| $Nu_a$ | Nusselt number associated with modified heat exchanger |
| $Nu_o$ | Nusselt number associated with ordinary heat exchanger |
| $Pr$ | Prandtl number |
| $\Delta P$ | Drop in static pressure, Pa |
| $Q$ | Heat flux, W/m$^2$ |
| $Re$ | Reynolds number |
| $S$ | Source term of energy |
| $T_b$ | Bulk mean temperature, K |
| $T_w$ | Wall temperature, K |
| $V$ | Velocity, m/s |
| $y$ | Twist ratio |
| $\rho$ | Density, kg/m$^3$ |
| $\mu$ | Dynamic viscosity, kg/m-s |

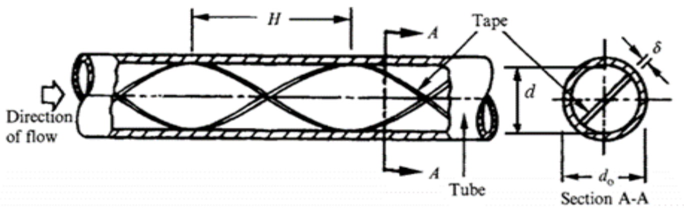



Fig. 1. Twisted tape insert

The overall enhancement ratio (OER), is a parameter which compares the heat transfer performance of an enhanced flow channel and an ordinary flow channel of similar geometry, under same pumping power conditions. The evaluation of OER involves Nusselt number (*Nu*) as heat transfer parameter and the friction factor (*f*) as pumping power or pressure drop parameter. Value of OER greater than or close to unity suggests that the enhancement technique is good.

Manglik and Bergles [1] conducted experimental studies in circular tubes with twisted tape inserts, with isothermal wall and turbulent flow conditions. They conducted experiments using ethylene glycol and water. They used tapes of twist ratios 3, 4.5 and 6. They also developed generalised correlations for Nusselt number and friction factor for transition and turbulent flow conditions. Patnala Sankara Rao and Kiran Kumar [2] conducted numerical analysis on a double pipe heat exchanger, with inner tube containing twisted tape of twist ratio 5 with water as working fluid. They compared the results of the experiment and numerical analysis with correlations proposed by Manglik and Bergles [1] and observed reasonable agreement between them. Hasanpour *et al.* [3] presented a review on the performance of twisted tape inserts on turbulent flow heat exchangers based on the overall enhancement ratio (OER) criteria. They considered various models of twisted tape and observed that all these tapes had OER greater than unity. Ahamed *et al.* [4] conducted an experimental investigation on a tube with perforated mild steel twisted tape inserts, with circular holes of different diameters. The twist ratio was taken as 4.55 for all tapes. Air was used as the working fluid. They observed the heat transfer enhancement by twisted tape with perforation is slightly higher than that of plain twisted tape of same twist ratio. Naga Sarada *et al*. [5] conducted experimental investigation on turbulent flow heat transfer in a horizontal tube by means of varying width twisted tape inserts with air as the working fluid. They found that the enhancement of heat transfer with twisted tape inserts as compared to plain tube varied from 36% to 48% for full width and 33% to 39% for reduced width inserts. The overall enhancement ratio of the tubes with full width twisted tape insert was found to be higher than that of reduced twisted tape insert. They conclude that material saving can be achieved by using twisted tapes of reduced width.

## 2. The proposed modification

Twisted tapes are conventionally used in the inner tubes of double pipe heat exchangers. When twisted tapes are inserted snug to loose fit inside the tube, it cannot produce any significant fin effect [1]. In this study, the enhancing element is a single Plain twisted tape (PTT) which affects the flow on both tube and annulus side of the heat exchanger. Insertion of twisted tape will induce swirl flow on both sides. The proposed geometry is shown in Fig.2.

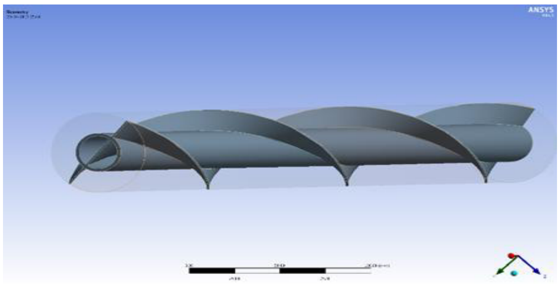



Fig. 2. The proposed modification

This modification is expected to result in better performance of the heat exchanger by enhancement of heat transfer coefficient on both sides and fin effect of twisted tape. From manufacturing point of view, such a geometry can be made by making two slits on the surface of inner pipe, starting diametrically opposite to each other through two helical path of equal pitch. A twisted tape of same pitch whose width is close to the inner diameter of the outer pipe with some fit allowance and thickness slightly less than the width of the slit made in the inner pipe, can be inserted in to the same. Then brazing or welding should be done along the contact surfaces of twisted tape and inner pipe to make it leak proof. The twist ratio is mentioned based on the inside diameter of the inner tube.

## 3. Data reduction

Local heat transfer coefficient associated with tube side or annulus side

$$h_l = \frac{Q}{T_w - T_b} \quad (2)$$

Nusselt number associated with tube side or annulus side

$$Nu = \frac{h_a D_h}{k} \quad (3)$$

where $h_a$ is the average heat transfer coefficient obtained by averaging local heat transfer coefficients on respective sides at eight equidistant sections considered along the length of the heat exchanger.

Fanning's friction factor

$$f = \frac{\Delta P}{2\left(\frac{L}{D}\right)\rho V^2} \quad (4)$$

Overall Enhancement ratio

$$\mathrm{OER} = \frac{Nu_a}{Nu_o} \Big/ \left(\frac{f_a}{f_0}\right)^{\frac{1}{3}} \quad (5)$$

For flow through tubes with twisted tape inserts, the correlations suggested by Manglik and Bergles [1] are used

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \left(\frac{\pi}{\pi - \frac{4\delta}{d}}\right)^{0.8} \left(\frac{\pi + 2 - \frac{2\delta}{d}}{\pi - \frac{4\delta}{d}}\right) \left(\frac{\mu_b}{\mu_w}\right)^n \left(1 + \frac{0.769}{y}\right)^{0.2} \quad (6)$$

n=0.18 (for liquid heating), 0.30 (for liquid cooling)

$$f = \frac{0.0791}{Re^{0.25}} \left(\frac{\pi}{\pi - \frac{4\delta}{d}}\right)^{1.75} \left(\frac{\pi + 2 - \frac{2\delta}{d}}{\pi - \frac{4\delta}{d}}\right)^{1.25} \left(1 + \frac{2.752}{y^{1.29}}\right) \quad (7)$$

$$Re = \frac{\rho V D_h}{\mu} \quad (8)$$

For flow through ordinary heat exchanger, following correlations can be used

$$Nu = 0.023 Re^{0.8} Pr^n \quad (9)$$

n=0.4 for heating and 0.3 for cooling

$$f = \frac{0.079}{Re^{0.25}} \quad (10)$$

## 4. Details of CFD analysis

The CFD analysis was carried out using the commercial CFD package ANSYS-FLUENT, version 14.5. Modelling software CATIA V5 was used to model the geometry of heat exchangers.

The present work is based on the following assumptions:

- 3-Dimensional, steady, incompressible flow
- Body forces are neglected
- Heat exchanger section is perfectly insulated from surroundings
- Property variations of fluids are negligible
- Radiation effects are neglected
- No heat generation within the fluid

The governing equations are:
Continuity equation

$$\nabla.(\rho\vec{v}) = 0 \tag{11}$$

Momentum equation

$$\nabla.(\rho\vec{v}\vec{v}) = -\nabla P + \nabla.(\mu\nabla\vec{v}) \tag{12}$$

Energy equation

$$\nabla.\left(\rho\vec{v}C_pT\right) = \nabla.(k\nabla T) \tag{13}$$

The SST k-$\omega$ turbulence equation for turbulent kinetic energy $k$ and specific dissipation rate $\omega$ in conserved form

$$\frac{\partial k}{\partial t} + U_j\frac{\partial k}{\partial x_j} = P_k - \beta^* k\omega + \frac{\partial}{\partial x_j}\left[(\nu + \sigma_k \nu_T)\frac{\partial k}{\partial x_j}\right] \tag{14}$$

$$\frac{\partial \omega}{\partial t} + U_j\frac{\partial \omega}{\partial x_j} = \alpha S^2 - \beta\omega^2 + \frac{\partial}{\partial x_j}\left[(\nu + \sigma_\omega \nu_T)\frac{\partial \omega}{\partial x_j}\right] + 2(1-F_1)\sigma_{\omega 2}\frac{1}{\omega}\frac{\partial k}{\partial x_i}\frac{\partial \omega}{\partial x_i} \tag{15}$$

Mesh independence study was conducted to select optimum mesh for different domains. The boundary conditions specified are as follows. Hot water is entering the inner tube at $80^0$C and cold water entering the annulus at $27^0$C in counter flow arrangement. In a particular model, same flow velocity was given to both fluids and the velocity of flow was varied simultaneously in both tube and annulus. The velocities considered are 0.26, 0.23. 0.20, 0.17, and 0.14 m/s. Corresponding Reynolds numbers are 14591, 12908, 11224, 9540, 7857 in the tube side and 8760, 7749, 6739, 5727, 4717 in the annulus side respectively. Pressure outlet boundary condition is assigned to outlets of both fluids assuming the fluids reach the outlet at atmospheric pressure.

## 5. Results and discussion

### *5.1. Geometric Parameters and Materials*

Table 2 shows the physical dimensions of the heat exchanger. Ordinary and modified heat exchangers have the same dimensions specified in the table, with the only difference of presence of twisted tape inserts in modified heat exchangers.

Table 2. Physical parameters of heat exchanger

| Parameter | Value |
|---|---|
| Length of heat exchanger | 800 mm |
| Inner diameter of inner tube | 22 mm |
| Outer diameter of inner tube | 26 mm |
| Inner diameter of outer tube | 54 mm |
| Outer diameter of outer tube | 58 mm |
| Thickness of twisted tape | 1 mm |
| Twist ratio of twisted tape | 3,5 |

Water is chosen as both hot and cold fluids. The properties of the fluids are evaluated at the mean of respective inlet and outlet temperatures. All the solid bodies, i.e. the inner tube, outer tube and the twisted tapes are formed out of Aluminium.

### *5.2. Validation and comparison of results*

For ordinary heat exchanger Nusselt number is validated using (9) and friction factor using (10). The Nusselt number and friction factor for tube side flow in double pipe heat exchanger with twisted tape inserts is validated using (6) and (7). Fig.3 shows the variation of tube side Nusselt number and friction factor with Reynolds number in various models of heat exchanger. Figs.3a and 3b shows the variation of tube side Nusselt number and tube side friction factor with Reynolds number.

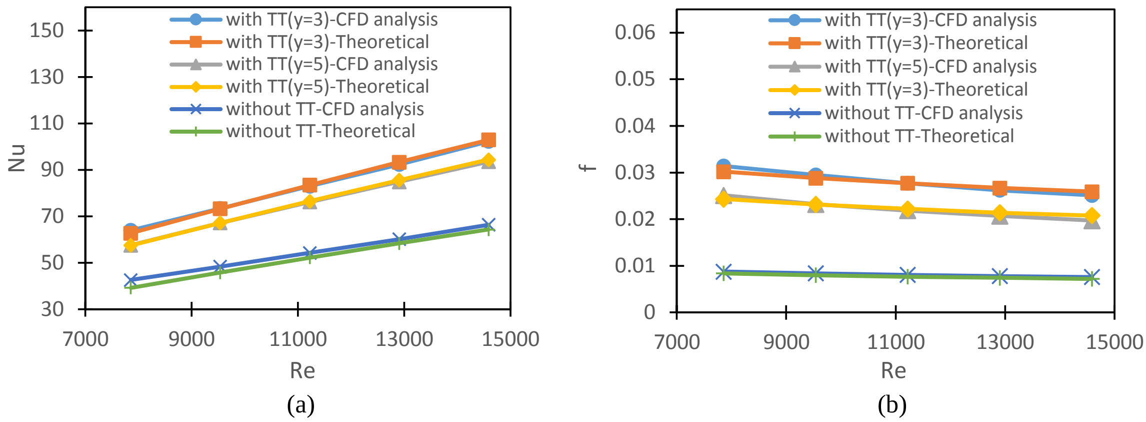



Fig.3. (a) Variation of Tube Side Nusselt Number with Reynolds number; (b) Variation of Tube Side Friction factor with Reynolds number

From Figs.3a and 3b, it can be seen that the values of Nusselt number and friction factor obtained by numerical analysis show good agreement with the results obtained from correlations

From Fig.3a, it is seen that twisted tape of twist ratio 3 enhanced heat transfer more than that with twist ratio 5. This is because twisted tape of twist ratio 3 have severe twist and increase the effect of swirl flow and mixing. The value of Nusselt number obtained in heat exchanger with twisted tape insert of twist ratio 3 and 5 are around 1.5 and 1.4 times that of the ordinary heat exchanger. From Fig.3b, it is observed that by the insertion of twisted tape, higher tube blockage is experienced and it increases further with decreasing twist ratios. The tube side friction factor of heat exchanger with twisted tapes of twist ratio 3 and 5 were almost 3.5 and 2.7 times compared to that of ordinary heat exchanger tube.

Fig.4.a shows the variation of tube side Overall Enhancement Ratio (OER) with Reynolds number. Fig.4b shows the variation of annulus side Nusselt number with Reynolds number in various models of heat exchanger

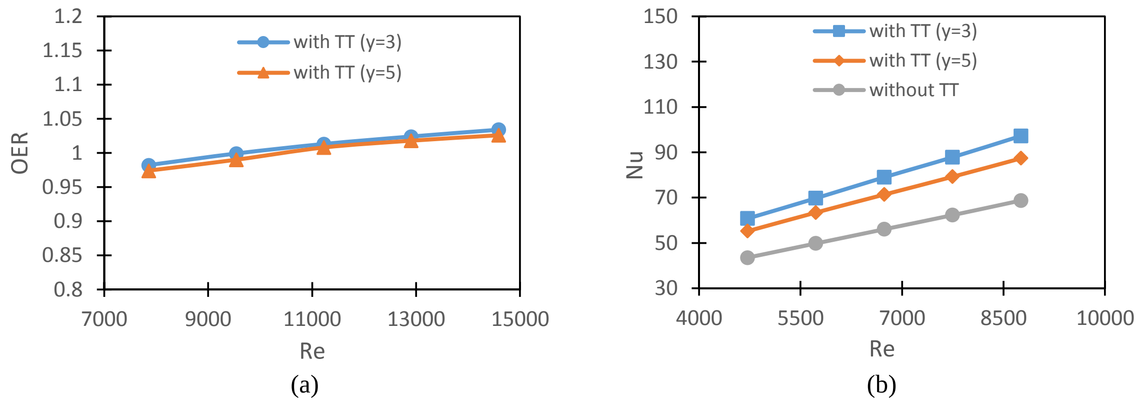
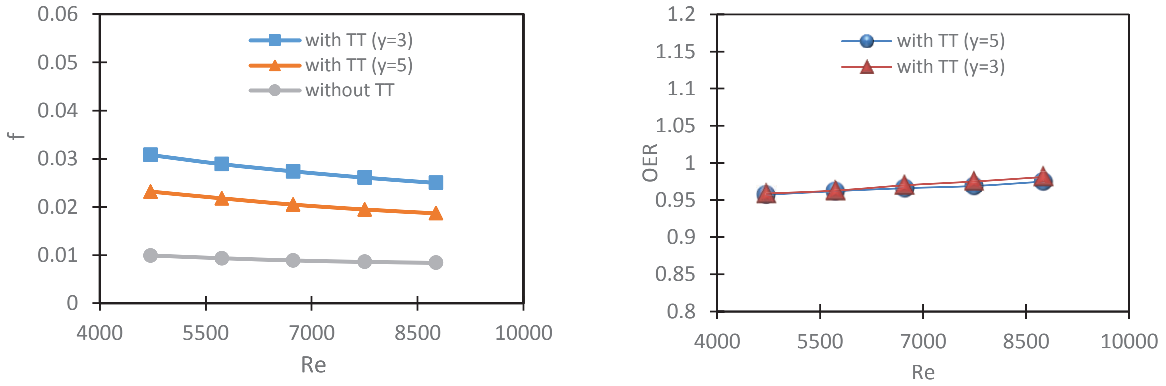



Fig.4. (a) Variation of tube side Overall Enhancement Ratio (OER) with Reynolds number for different twist ratios; (b) Variation of annulus side Nusselt number with Reynolds number in various models of heat exchanger

From Fig.4a, it is understood, that both twisted tapes have OER around unity when the tube side flow is considered. This shows that the penalty in pressure drop is justified by the increase in heat transfer. A trend of increasing OER with Reynolds number can be seen for both twist ratios. From Fig.4b, it is seen that twisted tapes considerably increased heat transfer. Nusselt number in annulus with twisted tape inserts were about 1.4 (with tape of twist ratio 3) and 1.3 (with tape of twist ratio 5) times greater than that of annulus of ordinary heat exchangers.

Fig.5a shows the variation of annulus side friction factor with Reynolds number in various models of heat exchanger. Fig.5b shows the variation of annulus side OER with Reynolds number for different twist ratios




(a) (b)

Fig.5. (a) Variation of annulus side friction factor with Reynolds number in various models of the heat exchanger; (b) Variation of annulus side Overall Enhancement Ratio with Reynolds number for different twist ratios

From Fig.5a it is understood that, the friction factor is highest in the annulus of heat exchanger with twisted tape of twist ratio 3. The friction factor in is about 3 times that of friction factor obtained in the annulus of ordinary heat exchanger. Friction factor associated with twisted tape of twist ratio, y = 5 was about 2.3 times of that obtained in ordinary heat exchanger. From Fig.5b it is seen that values of OER are again close to unity which justifies the performance of twisted tape inserts in the annulus also. As observed in the tube side flow, slight improvement in OER with Reynolds number is also evident.

### *5.3. Fin Effect of Twisted Tape Insert*

The single twisted tape which is used to enhance heat transfer in both sides of the heat exchanger, is a single continuous metal strip and is in contact with hot and cold fluids. This fact is expected to create a thermal circuit between the fluids through the twisted tape also. To analyse whether the fin effect produced by the twisted tape is significant or not, a model of heat exchanger was analysed by treating the walls of the twisted tape as perfectly insulated. Comparing the heat transfer in such a model with one operating at same flow conditions, but with conducting twisted tape, will reveal the fin effect of the twisted tape if there is any. A case with tube side flow of Reynolds number 12908 and annulus side flow of Reynolds number 7749 with a twisted tape of twist ratio 3, was selected such that in one case the twisted tape is conducting and in the other its surfaces are adiabatic. Fig.6 shows the heat flux on surface of the conducting twisted tape.

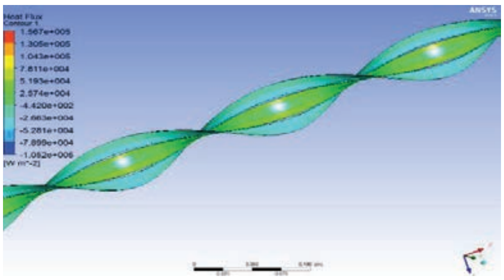


Fig.6. Heat flux on the surface of the twisted tape

Results shows that the heat transfer rate obtained when surface of twisted tape is treated as adiabatic, is about 21.5% less than the heat transfer rate obtained with conducting twisted tape condition. So it can be said that the twisted tape exhibit significant fin effect.

## 6. Conclusions

The major conclusions from the study can be concluded as follows:

- Insertion of twisted tape in double pipe heat exchanger improved the heat transfer coefficient on both tube side and annulus side of heat exchanger. Secondary flows induced by the twisted tape, enhanced cross stream mixing of the fluids, increase in the effective flow length and the fin effect of the twisted tape were the reasons behind improved performance of the heat exchanger.
- When the heat transfer and pressure drop characteristics were compared on the basis of "Overall Enhancement Ratio"(OER) criteria, results show that the twisted tape insert performed reasonably well both in tube and annulus, with OER around unity
- The attempt to study the fin effect produced by the twisted tape in the modified heat exchanger revealed that the twisted tape exhibit significant fin effect which is an advantage not covered by the OER criterion.
- With the additional benefit of fin effect of twisted tape and good OER values, we can conclude that the heat transfer enhancement dominates over the pressure drop penalty incurred.

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